Effect of Clearance Gap in Spiral Casing Design of a Centrifugal Fan with Optimized Impellers

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Abstract—Industrial fans are subject to European Union energy labeling and Ecodesign requirements. By using more efficient industrial fans, Europe will save 34 TWh and avoid 16 million tons of CO₂ emissions annually by 2020 [1]. In this paper, the effect of the clearance gap between the impeller and the volute, on the performance of a centrifugal fan was investigated using open source CFD software OpenFOAM [2]. An automated loop with RANS and data post-processing is set up using Matlab, for allowing a large number of parameter variations. We conducted numerical analysis for all operating points, where starting points are optimal impellers for the whole range of specific speeds [3], [4]. The effect of volute angle and geometrical parameters related to the tongue [5], on total pressure loss, static pressure recovery coefficient and on efficiency are presented.

Index Terms—CFD, Clearance Gap, Efficiency, Volute.

I. INTRODUCTION

Many articles related to centrifugal fans have studied and optimized only the fan impeller and some of them treat the fan as a whole unit, while the study of the spiral casing is less well-known. From the literature it is known that the spiral casing can take large part of the hydraulic losses in the fan. Minimization of energy loss depends on the characteristics of the spiral casing. Hence, appropriate design of the fan spiral casing has significant meaning to centrifugal fan performance.

For this reason, this study of the effect of the clearance gap has been conducted, which should lead to better advanced recommendations for the shape of the spiral casing.

II. VOLUTE SHAPE DESIGN METHOD

Constant circulation method [6] is a method applied by drawing a spiral case based on the fact that velocity circulation is a constant \( r_c = \text{constant} \). In practice, this rule is valid with the restriction that one spiral must be so far displaced from the impeller that deflections conditioned by the consideration of a finite number of blades can be ignored. This rule constitutes the basis for the dimensioning of a volute in the cases where friction has been ignored. The velocity \( c \) at an arbitrary place can be calculated from its components \( c_m \) and \( c_u \), \( r c_u = r_2 c_u \). From the condition that the same volume-flow must flow (the continuity equation) through all the streamline in volute it gives the correlation:

\[
Q = 2\pi r_2 b c_m = 2\pi B c_m
\]  

(1)

From which follows \( r_2 b c_m = r B c_m \), from this we obtain the following inclination \( \alpha \) of the streamlines:

\[
tg(\alpha) = \frac{c_m}{c_u} = \frac{c_{m2} b_2}{c_{u2} B}
\]  

(2)

Because we obtain the boundary of the volute from the streamline, again it yields, \( tg(\alpha) = \frac{dr}{r d\varphi} \)

\[
\frac{dr}{r} = d\varphi \cdot tg(\alpha) = d\varphi \cdot tg(\alpha) \frac{b_2}{B}
\]

(3)

The solution states,

\[
\ln \frac{r}{r_2} = \varphi \cdot tg(\alpha) \frac{b_2}{B} = \varphi \cdot \frac{c_{m2} b_2}{c_{u2} B}
\]  

(4)

Accordingly, the trajectory of fluid particles in the volute is as follows (Carolus 2013) [7],

\[
r(\varphi) = r_2 e^{\varphi tg(\alpha)} = r_2 e^{\varphi^t g(\alpha) b_2 \frac{B}{B}}
\]  

(5)

\( r(\varphi) \), is the radius of the volute at an angle \( \varphi \), \( r_2 \), is the outer radius of the impeller that is equal to 150mm in our case.

\( \alpha \), is the angle that the absolute velocity vector makes with the peripheral direction \( tg(\alpha) = c_m/c_u \).

\( b_2 \), the width of outlet impeller \( B \), the width of volute

Fig. 1. Geometry parameters of the spiral casing (Carolus 2013) [3]

In this study, is shown the effect of clearance gap on the performance of the centrifugal fan by using open source CFD software OpenFOAM. A qualitative understanding of the effects of parameters will enable the performance of a real product to be improved.
III. PERFORMANCE OF THE VOLUTE

The overall performance of the volute is affected mainly by the following geometric parameters [5]:
- area of the cross-section,
- the shape of the cross-section,
- radial location of the cross-section,
- location of the impeller and
- tongue geometry

The overall performance of the volute can be analyzed by using:

Total pressure loss coefficient of volute:

\[
K_p = \frac{p_{t2} - p_{t3}}{p_{t2} - p_2} = \frac{p_{t2} - p_{t3}}{\frac{V}{2}\rho c_2^2}
\] (6)

\(K_p\) is defined as the ratio between the total pressure losses in the volute to the dynamic pressure at the impeller exit.

Static pressure recovery coefficient of volute:

\[
C_p = \frac{p_3 - p_2}{p_{t2} - p_2} = \frac{p_3 - p_2}{\frac{V}{2}\rho c_2^2}
\] (7)

\(C_p\) is defined as the ratio between the static pressure recovered in the volute to the dynamic pressure at the impeller exit.

Total efficiency of volute

\[
\eta_T = \frac{p_{t3}}{p_{t2}}
\] (8)

From equation (6, 7) becomes:

\[
C_p = 1 - K_p - \left(\frac{c_3}{c_2}\right)^2
\] (9)

\(\left(\frac{c_3}{c_2}\right)^2\) is the ratio of volute outlet/inlet kinetic energy.

Maximizing performance of a complete fan requires,

finding \(c_{3,\text{opt}}, K_{p,\text{opt}} = K_{p,\text{min}}, C_{p,\text{opt}} = C_{p,\text{max}}\) from CFD as a function of clearance gap of volute (function of alpha spiral angle, tongue angle and tongue radius).

IV. NUMERICAL ANALYSIS

The detailed flow field at the impeller’s outlet from preceding RANS simulations is used as boundary conditions for a RANS of the flow in the volutes.

Three-dimensional, incompressible, steady-state flow simulations were performed using the Open Source CFD software, OpenFOAM v3.0.x [2]. This solves discretized forms of the Reynolds-averaged Navier–Stokes equations for turbulent flow using the finite volume method (Ferziger, Perić 2002) [8].

The unstructured grid solution procedure is based on a variant of the SIMPLE pressure correction technique (Patankar 1980) [9]. The iterative solution was deemed to be converged when the normalized absolute error over the mesh had reduced to 10^{-5} for each variable. OpenFOAM supports the standard k-\(\omega\) model by Wilcox (1998) [10], and Menter’s SST k-\(\omega\) model (1994) [11]. The k-\(\omega\) SST turbulence model was employed for these calculations, with near-wall conditions supplied by the 'wall function' conditions of Launder and Spalding, 1974 [12].

V. BOUNDARY AND INITIAL CONDITIONS

The inflow boundary conditions were based on known flow rates and the flow direction [3], [4]. Nonuniform velocity profiles were prescribed at the volute inlet (fan impeller outlet) by implementing radial and tangential velocity components, also axial velocity is included. The front and backside of the impeller as the rotating wall, the other parts wall with the no-slip condition and for the outlet ambient pressure is used. Turbulent kinetic energy is \(k = 3m^2s^{-2}\), and the specific turbulence dissipation rate is \(\omega = 4000s^{-1}\).

The geometry of volutes is generated from MATLAB as stereolithography (.stl file) than cfMesh v1.1.2 software is used to create a mesh. The grid resolution is made according to \(y^+\) value 30 < \(y^+\) < 200.

VI. CFD SIMULATION RESULTS

This study will focus on the effect of the clearance gap between the rotor and the spiral casing. In our previous study [4], the influence of parameters is given but separately and not as a function of the clearance gap. Considering the design of the spiral casing, clearance gap is a function of the logarithmic spiral, \(\alpha\), the angle of the tongue \(\phi_x\), and the radius of the tongue \(r_2\). Since we will study every operation point of the fan, the angle of spiral casing will vary from \(\alpha = 12 \div 20^\circ\). Width of the spiral casing is accepted constant since this parameter does not affect the clearance gap value, see Fig. 1. Referring to the literature we will have references to the values of these parameters that can obtain [13]. To observe the effect of each parameter, we will first accept some constant parameters and then continuously examine the other parameters. We will assume an average alpha angle value \(\alpha = 16^\circ\) and the angle of the tongue \(\phi_x = 35^\circ\).

We will change the radius of the tongue to the values \(r_2/D_2 = 2.5 \div 5\%\). The results are as follows:

![Fig. 2. Effect of tongue radius on the total efficiency of spiral casing](image)

What we can identify is that changing the radius of the tongue of the spiral casing has no effect on the total efficiency for each of operating point. The maximum
efficiency achieved in a spiral casing refers to the flow coefficient $\phi = 0.13 \div 0.14$. We need to look at the effect on the static pressure recovery coefficient of the spiral casing.

We obtain the same result for the static pressure recovery coefficient in the spiral casing. But unlike the efficiency of the spiral casing, the static pressure recovery coefficient gets the maximum value for the minimum flow coefficient and the positive value for the flow coefficient from $\phi = 0.07 \div 0.12$.

Since the radius of the tongue has no effect on the efficiency then we will consider other parameter values of the spiral casing tongue. To ensure a minimum safety clearance distance of $\frac{S_z}{D_2} = 3.5\%$ [13] it has been accepted as a minimum tongue angle of $\varphi_z = 35\%$. The results are as follows:

From Fig. 4, we observe a continuous increase in efficiency as the flow coefficient increases, and this trend is the same for each tongue angle value. Except with increasing the angle of the tongue $\varphi_z$, we will decrease the total efficiency of the spiral casing.

Fig. 3. Effect on the static pressure recovery coefficient of the spiral casing function of the radius of the tongue.

Fig. 4. Total efficiency of the spiral casing as a function of flow coefficient

If we observe the static pressure recovery values, we can see that for the minimum flow coefficient, the maximum pressure recovery values are obtained. In this case, with the increase in the angle of the spiral casing tongue, there is also an increase in the values of the static pressure recovery coefficient. The optimum maximum value is obtained for the flow coefficient of $\phi = 0.07 \div 0.08$. If we refer to the large values of the flow coefficient, negative values of the static pressure recovery coefficient are also observed.

However, to have a better understanding of the behavior of each parameter, we must also see at the total pressure loss coefficient.

Since for the total pressure loss coefficient is required a value as small as possible, therefore the optimal values are in the range of flow coefficient from $\phi = 0.10 \div 0.11$. This is the recommended working area of this type of fan with this type of spiral casing. If we move beyond this work area, the value of pressure losses will be higher. It is important to note that for the flow coefficient $\phi = 0.07 \div 0.10$, the minimum value of $K_p$ is for tongue angle $\varphi_z = 60\%$, whereas for $\phi = 0.11 \div 0.14$, the minimum value of $K_p$ is for $\varphi_z = 35\%$.

Another parameter that influenced the safety space between the rotor and the spiral casing is the logarithmic alpha angle. We have accepted 2 limit values, for compact spiral casing we have chosen 12-degree alpha angle (providing minimum security space) and for large spiral casing 20-degree angle. Since the alpha angle directly affects the spiral casing size, if we increase the alpha angle we expect to have a low-efficiency value and an increased value in the static pressure recovery coefficient. In order to determine the optimal working point of the spiral casing, we must refer to the values of the total pressure loss coefficient.

Fig. 5. Static pressure recovery coefficient as a function of flow coefficient

Fig. 6. Total pressure loss coefficient as a function of flow coefficient
For each of the logarithmic spiral alpha angles, we have a minimum optimum value of the total pressure loss coefficient. And in this case, the recommended values of the working point are in the range $\phi = 0.09 \div 0.12$.

To verify this value another parameter called the ratio of the output/input kinetic energy $\left(\frac{c_3}{c_2}\right)^2$, must be considered, where it must be ensured that this ratio is less than 1. The graph below shows that for the alpha angle 12 degree, the ratio $\left(\frac{c_3}{c_2}\right)^2$ is less than 1 for the flow coefficient value of $\phi < 0.12$.

Since in this study we presented the effect of 3 different parameters, in particular, the effect of the clearance gap has to be shown as a parameter which is a function of the above three parameters $s_z = f(\alpha, r_z, \phi_z)$.

If we have a small value of clearance gap between the rotor and the spiral casing, we will have high values of total efficiency, but we cannot go beyond the value of 3.5% due to security reasons [14]. If we refer to the flow coefficient, with its increase we will have an increase in the total efficiency of the spiral casing.

Since the recommended values of the flow coefficient were $\phi = 0.09 \div 0.12$, we are only presenting the results below for these values.

If we notice the recommended interval of fan working point for this type of spiral casing, it is observed that for the flow coefficient of $\phi = 0.09$, the minimum value of $K_p$ is in the range $\frac{s_z}{D_2} = 4.5 \div 5\%$. For the value of the flow coefficient of $\phi = 0.10$, the minimum value is reached for clearance gap around $\frac{s_z}{D_2} = 5.25 \div 5.75\%$, whereas for the flow coefficient of $\phi = 0.11 \div 0.12$, the smallest value of the total pressure loss coefficient is observed for clearance gap around $\frac{s_z}{D_2} = 6.6 \div 7.5\%$.

VII. CONCLUSION

From the results of numerical simulations, we come to the following conclusions:

- The radius of the tongue has no effect on the total efficiency, on static pressure recovery coefficient and on total pressure loss coefficient of the spiral casing.
- Maximum efficiency is achieved for flow coefficient $\phi = 0.13 \div 0.14$, whereas the maximum static pressure recovery coefficient is achieved for values of flow coefficient $\phi = 0.07 \div 0.12$.  

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• Increasing the angle of the tongue will decrease efficiency whereas increasing the static pressure recovery coefficient in the spiral casing.
• If we refer to the spiral casing tongue angle, the recommended working point for this fan type and for this spiral casing type is for flow coefficient $\phi = 0.10 \div 0.11$.
• If we refer to the angle of the spiral casing, the working point recommended for this fan type and for this spiral casing type is for flow coefficient $\phi = 0.09 \div 0.12$.
• If we refer to the effect of clearance gap as a function of three parameters $s_2 = f(\alpha, r_p, \varphi_z)$, we will have:
  - for the flow coefficient $\phi = 0.09$, the minimum value of $K_p$ is in the range $\frac{s_2}{D_2} = 4.5 \div 5\%$
  - for the flow coefficient $\phi = 0.10$, the minimum value of $K_p$ is in the range $\frac{s_2}{D_2} = 5.25 \div 5.75\%$
  - For the flow coefficient $\phi = 0.11 \div 0.12$, the minimum value of $K_p$ is in the range $\frac{s_2}{D_2} = 6.6 \div 7.5\%$.

NOMENCLATURE

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ABBREVIATIONS

- CFD: Computational Fluid Dynamics
- RANS: Reynolds Averaged Navier-Stokes
- SST: Shear stress transport
- FOAM: Field Operation And Manipulation
- SIMPLE: Semi-Implicit Method for Pressure Linked Equations

REFERENCES


Ardit GJETA was born in Peshkopi, Albania in 1988. He received his M.Sc. degrees in mechanical engineering from Polytechnic University of Tirana in 2012.
Since 2013, he joined Polytechnic University of Tirana as a lecturer in the energy department, where he teaches subjects like thermodynamics and fluid machines. He is currently following the PhD studies.
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